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ADDED PROPERTIES EFFECT IN HYDRAULIC TURBINES

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The interactions between a structure with its surroundings are important. A realistic prediction of the dynamic behavior of hydraulic turbines in different design conditions is very important for a reliable operation. These interactions are most important when the surrounding fluid is dense. In the design of the hydraulic turbines the dynamic response of the runner must be taken into consideration.

The paper aims to present the research done so far in this domain starting with the first analysis done for simple bodies and geometries and continuing with the research done in the hydropower field. An experimental procedure to analyze the added properties in case of a Francis hydraulic turbine is presented together with the results for the analyzed case. The values obtained for added properties led to the conclusion that the procedure gives significant information, and that further investigation must be performed for various operating conditions.

Keywords: added properties, hydropower, hydraulic turbines.

1. Introduction

In the last years, the emphasis on renewable energies increased significantly. Technologies in this field recorded a significant improvement, leading to the use of large-scale wind and solar parks for electricity production, which introduced an increased number of fluctuations on the grid. The deregulation of electricity markets, coupled to the increased use of intermittent

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renewable energy sources to the grid, resulted in an increase of the hydropower use to maintain grid stability. Hydraulic turbines are designed for steady state conditions. Frequent load variations and large number of start-stop cycles may cause vibration, fatigue, wear and changes in the dynamic behavior of hydraulic turbines [1]. The new hydraulic turbine design methods must take into account all these new problems arising in operation. Nowadays, great emphasis is on refurbishment of these units in order to increase performance and enlarge the optimal functioning [2]. These can be achieved considering the turbine control system through the governor and a part of rotor dynamics.

Turbine speed governor plays a very important role in hydraulic transients caused by load changes and grid instabilities. Hydroelectric power generating system exhibits a high - order and nonlinear behavior. Appropriate mathematical models have the essential tools for simulation of such systems and to better overcome the transient regimes. The effect of a surrounding fluid on the turbine runner represents an important parameter of the system. The controller must take into consideration the effect of the water given to the runner because the dynamics of the runner are influenced in a great part by this effect. This is important in steady as well as transient regimes. The effect is synthesized by taking into consideration additional terms into the motion equation of the runner. The water effect influences the inertia, the damping and the stiffness of the runner-shaft-generator system.

Identifying this effect could improve the operation of hydraulic turbines and could avoid problems that appear, like vibrations. In the last years a small interest of this effect was considered, mostly on simple bodies, like cylinders. A complete analyze for a hydraulic turbine was not completed yet. Experimental and simulated results of this kind of analyze are needed at this moment.

The effect of a surrounding fluid on the natural frequencies and mode shapes of a structure is not ordinarily significant for relatively compact structures if the fluid density is much less than the average density of the structure. Thus, the surrounding air does not usually affect the natural frequencies or mode shapes of most metallic structures. The added mass accounts for the inertia of the fluid entrained by the accelerating structure. As the structure accelerates, the fluid surrounding the structure must accelerate as well. The inertia of the entrained fluid is the added mass.

This paper aims to present the research done so far in this field, presenting the first theoretical and experimental results obtained for simple geometries and bodies. Also the procedure described here is a proper way to quantify the added properties effects, in the case of a Francis turbine laboratory model. The first results of the analysis are being presented.

2. General research on added properties

The concept of "added mass" was recognized starting from 1779, when Chevalier Du Buat [3] experimented it with spheres oscillating in water. After that, in 1828, Bessel [3] conducted experiments on spherical pendulums both in air and in water. It was noticed that the object's period of motion in water differed from that in air, and both of them concluded that it is necessary to attribute to the test object a virtual mass greater than its ordinary mass. The concept of "added mass" was thus introduced.

Determination of added mass by experiment, however, has often been inconsistent and unreasonable. Added mass is a variable that depends on the state of motion and is not a constant as had been shown by studies of potential flow [4].

Added mass can be expressed in many ways. In classical formulations, the hydrodynamic force and moment acting on a rigid body in a non-rotational flow of an unbounded non-viscid fluid are expressed in terms of the added mass coefficients of the body and the components of its velocity and acceleration [5]. In other cases, the added mass coefficients could be obtained through the use of the velocity potential.

From the basic theory, the motion equation of a simple supported symmetric body vibrating in a still fluid is:

$$J\ddot{\Phi} + C\dot{\Phi} + K\Phi = T \tag{1}$$

where J is the inertia of the body, C is the structural damping, K is the stiffness, T is the external torque applied to the body and Φ is the displacement of the body from the equilibrium position. For the case of unsteady motion of underwater bodies or unsteady flow around objects, it must be considered the additional effect resulting from the fluid acting on the structure when formulating the system equation of motion:

$$\left(J + J_w\right)\ddot{\Phi} + \left(C + C_w\right)\dot{\Phi} + \left(K + K_w\right)\Phi = T \tag{2}$$

where J_w, C_w and K_w are the new terms which influence the entire system, due to water interaction.

Stelson and Mavis [6] determined the added masses for the first mode frequencies of cylinders, spheres, and rectangles suspended in air and in water from a flexible support and set into free vibration. An evaluation for obtaining the added mass and added damping for a circular cylinder oscillating in linearly stratified fluid is being conducted by Ermanyuk [7], who showed the effect of the density stratification over the frequency dependent hydrodynamic coefficients like the added mass and added damping. Based on the equations developed by Cummins [8], Perez and Fossen [9] provided values for added mass and damping of marine structures in waves. McConnell and Young [10] investigated the dependence of added mass and damping on Stokes number, for a sphere in a bounded fluid. Based on the general agreement between theoretical and experimental results, McConnell and Young concluded that the Stokes viscous flow solution would accurately predict the added mass and fluid damping coefficients for spheres. Using the theory for the vibrating chord and considering that the membrane is vibrating in vacuum and the air movement around it is non-rotational and non-viscid, Yuanqi Li [11] obtained a relation for the added mass, depending only on the air density and chord length.

3. Added properties on hydraulic machinery

Over the last few years, investigations on turbine dynamic were carried out mostly through numerical simulations. The development of new measuring instruments and techniques allowed new types of measurements. The research for added properties was mostly done for added inertia and was based on comparison of the natural frequencies in air and water.

Rodriguez [12] tried an approach to determine the added mass effects using a reduced scale model of a turbine runner. A series of tests were conducted in air and in water. The response was measured with different accelerometers put in various positions on the runner. The natural frequencies and damping ratios and also mode-shapes were determined in air and water. The main conclusion of the experiment was that the difference in the natural frequencies is based only on the added mass effect and almost none on added damping. This difference depends also on the geometry of the mode, and has different values for different mode-shapes.

The mechanical design of hydraulic turbines is conditioned by the dynamic response of the runner that is usually estimated by a computational model. Liang and Rodriguez [13] aimed to obtain the effect of a given volume of water that surrounds the Francis runner by finite element method. The numerical model was based on the dynamic equation and the acoustic wave equation known as the Helmholtz equation. Certain simplifications were also considered, like easy compressible fluid, non-viscous, and the changes of density and pressure are very small. Added inertia effect was determined by comparing the results obtained for the natural frequencies of the simulations for water and air. Calculated frequencies in the two cases were compared with the experimental ones. Mode shapes between air and water were about the same even if frequency decreases significantly. Maximum displacement was found in areas of the blade edge. The results of the analysis are presented below, Fig.1, where the reduction of the natural frequencies from air to water is observed, with simulated and experimental results.

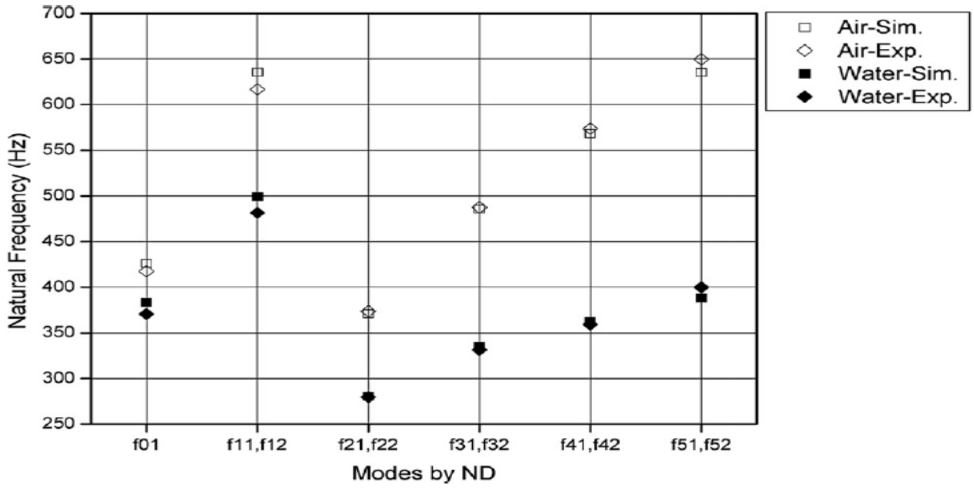


Fig. 1. Comparison of frequency reduction ratios obtained by simulation and experiment [13]

A similar analysis was conducted for a Kaplan turbine runner by de Souza Braga [14]. The response was measured also with accelerometers in different positions of a runner blade. The tests were made in free air and natural frequencies, damping ratios and mode-shapes were determined. Using also finite element method a numerical simulation was carried out to analyze the influence of the surrounding water in a turbine runner. The added mass effect was obtained by comparison between the natural frequencies and mode shapes in free air and in water. These results compared with some experimental results that were available showed a good agreement.

Using data from a hydropower plant, Salajka and Feilhauer [15] analyzed the natural frequencies of the runner-shaft-generator system in both water and air using finite element method. Natural frequencies and modes of vibration of the rotor are computed and consequently they were correlated with natural frequencies for all considered environments (vacuum, water, air). They obtained the reduction coefficients due to water effects.

Karlsson and Nilsson [16] showed that the behavior in terms of rotor dynamics is influenced by the interaction between fluid and rotor. The considered mechanical model is presented in Fig.2.

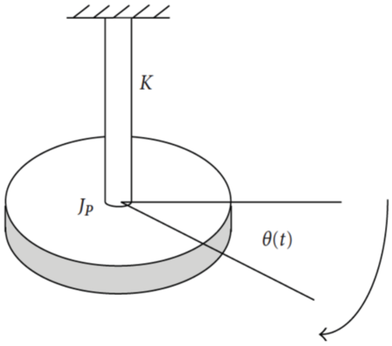


Fig. 2. Mechanical model of a torsional dynamic system: K – stiffness; J_p – polar moment of inertia; $\theta(t)$ – angular displacement [16]

Through the CFD Tool, inertia, damping and stiffness coefficients were estimated for a Kaplan turbine rotor with five blades. In the present work the rotational speed of the turbine is prescribed in order to determine the dynamical coefficients of the turbine runner due to flow. Simulations were made for the three loading of the turbine cases: 35, 60 and 70 %. To obtain the coefficients for the motion equation a simple model was built in which the generator is rigid and thus only the rotation of the rotor was considered. The results showed that these coefficients varied with the load of the turbine, the frequency resonance dropped with 5-65 % and the damping increased with 30-80 %. The results of the analysis, which vary with the frequency of the perturbation, are presented in Fig.3.

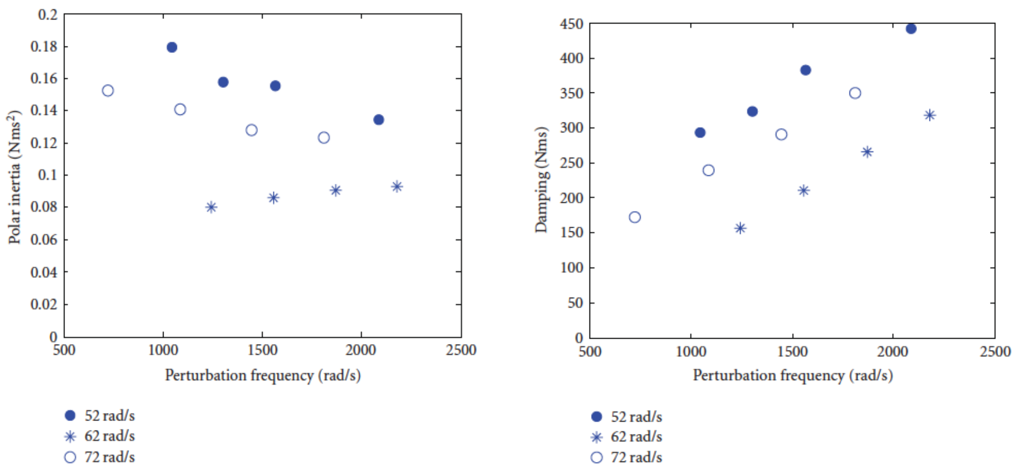


Fig. 3 Added inertia and damping [16]

With the increase of the perturbation frequency, a decrease into the added inertia is observed, along with an increase in the added damping values.

Most of the analyses done so far in this field were numerical simulations and the measurements done were only vibrations and mode shape measurements.

4. Experimental procedure for added properties determination

Currently, to conduct measurements in order to analyze the added properties effect is still a difficult thing to do. The controller and the programmable logic controller (PLC) of the hydropower unit modify and cancel any variation of the operating parameters. The most convenient procedure to do this kind of measurements is to conduct them in a laboratory test rig.

At the Norwegian University of Science and Technology (NTNU), Trondheim, Norway, in the Water Power Laboratory is a Francis turbine test rig that is properly equipped for this kind of measurements. The turbine represents a scaled-down model of the prototype operating at one of the most important hydropower plants in Norway, Tokke Power Plant. During the experiments in the laboratory, the model had at BEP an operating discharge of 0.2 m^3/s, a head of 12 m and a power output of 0.03 MW. The runner diameter of the model is 0.349 m. The rig can be used in both closed and open loop. Figure 4 presents the hydraulic circuit of the laboratory.

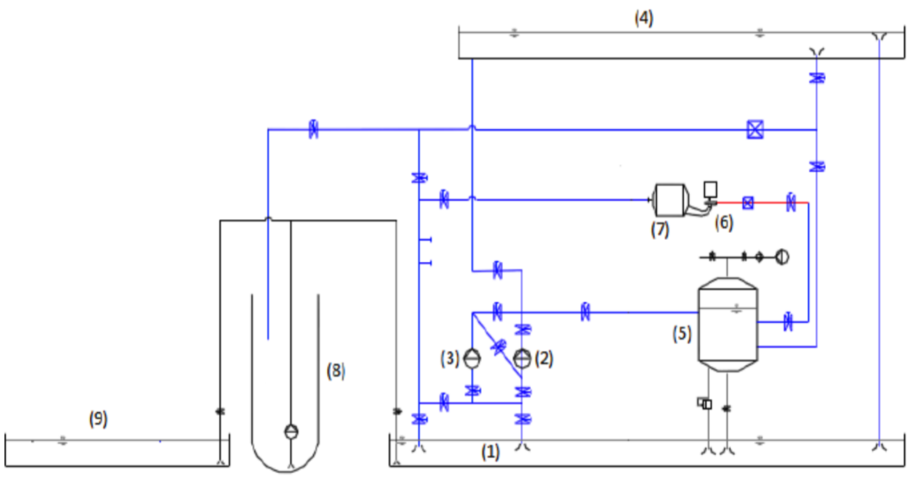


Fig. 4. Francis turbine test rig: (1, 9)-basement reservoir; (2, 3)-pumps; (4)-overhead tank; (5)-upstream pressure tank; (6)-Francis model turbine; (7)-downstream tank; (8)-weighting tank for flow meter calibration

The control of the turbine is done through the PLC. The control loop is presented in Fig. 5.

The procedure described in the following paragraphs is implemented from Karlsson [16], who made a similar approach using CFX simulations.

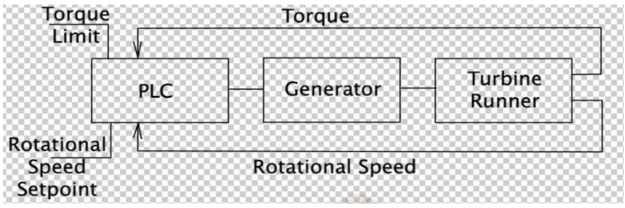


Fig. 5. Francis turbine control

There are two ways to control the rig: torque control and rotational speed control. Introducing a variation in the torque set point will create a variation in the rotational speed. Different variations in the torque set point could be created, like sinusoidal, step or impulse.

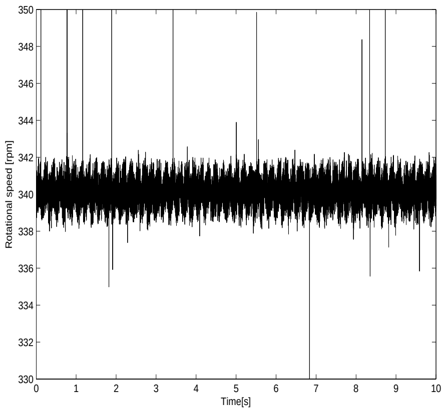
The measurements were made maintaining a constant power output at the generator. In the first stage of the experiment the torque limit is set to the maximum value that can be achieved by the generator. Modifying this value to the current torque value that is currently developed by the turbine and decreasing the rotational speed set point, the torque will have the tendency to increase but will be kept at the limit value. The perturbation introduced in the system is made by varying this limit in accordance to a sinusoid. Introducing the perturbation in the torque limit, the rotational speed will vary as well, along with the torque, creating the transient in the system. The perturbations introduced in the system may have different amplitudes, depending on the torque value of operating point and different frequencies as well.

All the calibrations and the measurements were conducted according to IEC TC 4 guidelines standards. A load cell, connected to an arm coupled to the generator casing, measured the generator torque. The angular speed was measured using a photocell and a circular disc with 60 cuts. The signals from the torque sensor and the optical fork sensor for angular speed were recorded using a data acquisition module, with a sample rate of 3012 Hz. The perturbation introduced in the system was controlled through a 4-20 mA signal that was created with NI CompactRIO and a current output card.

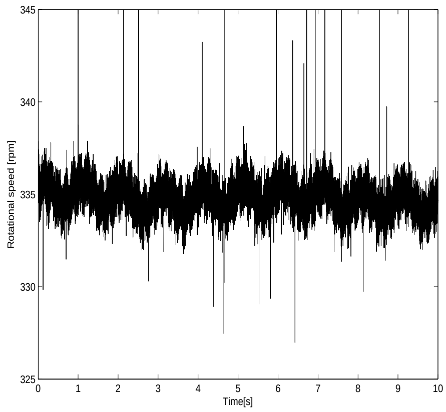
The measurements were carried out at the best efficiency operating point that was chosen from the hill diagram of the Francis turbine. At the best operating point the nominal torque was about 525 Nm and an angular speed of 334 rotations per minute (rpm). The perturbation was in the form of a sinusoidal with 1 Hz frequency and 3% amplitude of the torque value at the operating point.

5. Results

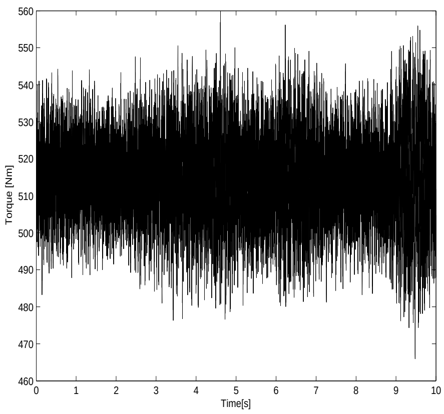
Figure 6 presents a comparison between the signals from the steady operating regime and the perturbation mode, for angular speed and mechanical torque signals. It can be seen that the form of the signal variation has a sinusoid shape in the perturbation regime (Fig. 6. b, d). Also, the frequency of the perturbation can be observed by measuring the time length between two peaks of the sinusoid, 1 cycle per second.



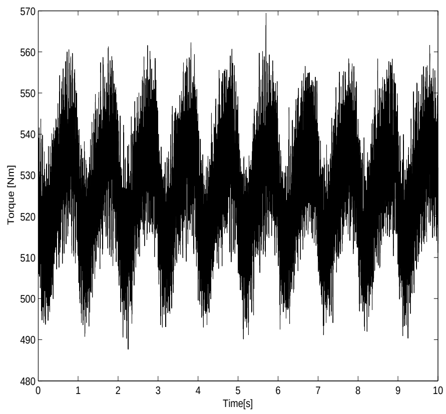
(a) Angular speed time domain signal – steady regime



(b) Angular speed time domain signal – perturbation regime



(c) Mechanical torque time domain signal – steady regime



(d) Mechanical torque time domain signal – perturbation regime

Fig. 6. Analyzed raw signals

As the signals are very noisy, the next step of the analysis consists in filtering them. The filter extracts only the signal component that contains the perturbation frequency, in the case presented 1 Hz ± 0.1 Hz. In the following figure, Fig.7, the time variation of the rotational speed and torque signal are presented, after filtering. The signals are zero-padded and overlapped, for better presentation.

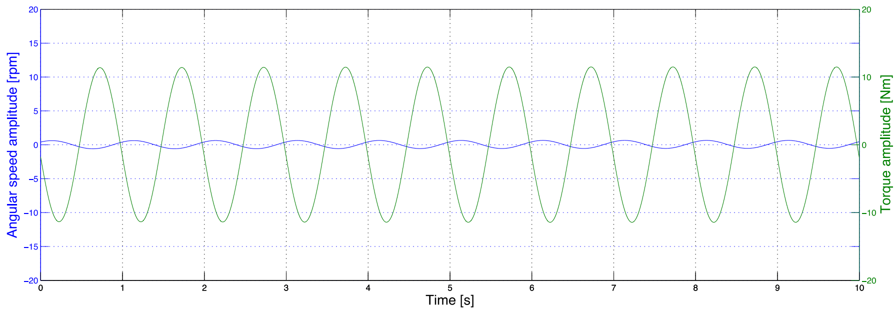


Fig. 7. Time domain filtered angular speed and mechanical torque signal

The amplitudes of the filtered signals are different, the one from the mechanical torque having higher amplitude (fig. 7). Also it can be seen that there is a phase shift between the two signals. Using FFT (Fast Fourier Transform), the amplitudes and the phase shift between the two signals can be captured. The amplitudes obtained here are, for the mechanical torque 11.35 Nm and for the angular speed 0.59 rpm.

Assuming that the stiffness is zero, and considering the angular displacement ϕ:

$$T = \tau \sin(\omega t) \tag{3}$$

the equation of motion becomes:

$$J\ddot{\Phi} + C\dot{\Phi} = \tau \sin(\omega t) \tag{4}$$

Considering a harmonic solution:

$$\phi = A \sin(\omega t) + B \cos(\omega t) = R \sin(\omega t + \alpha) \tag{5}$$

The solution becomes:

$$R = \frac{\sqrt{(\omega J)^2 + C^2}}{\omega^3 J^2 + \omega C^2}\tau \qquad \tan(\alpha) = \frac{C}{\omega J} \tag{6}$$

where α represents the phase shift between the two filtered signals.

Combining the equations above and the results from the measurements values for J and C are obtained:

$$J = 0.628 \text{ Nms}^2$$

$$C = 2.367 \text{ Nms}$$

6. Conclusions

The added properties are an important aspect in turbines operation nowadays and they need to be taken into consideration in the design stage and also in the operating mode. The paper presents the state of the art for the added properties topic, starting with the first experiments done for simple bodies and geometries, continuing with the research done in the hydropower field, along with an experimental procedure to determine them.

In the first step of the analysis, the experimental procedure implemented in the NTNU laboratory presents good results that will lead to find the added properties, for a Francis hydropower unit. Further research will be conducted for obtaining the added properties in different operation conditions and for characterizing the dynamic behavior of the turbine.

Acknowledgments

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